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Evaluation of on-machine measuring method for dynamic stiffness of thin-walled workpieces

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Abstract

Machining of thin-walled workpieces is a popular issue due to its static and dynamic displacements during machining. It is required to set machining conditions sufficiently considering the vibration characteristics of the workpiece. An automatic on-machine system to measure the dynamic stiffness of the workpiece enables a reliable and frequent measurement. In this study, an on-machine measurement device for workpiece dynamic stiffness is evaluated. The cause of the difference in natural frequency between the on-machine measurement method and impact test is investigated. The influence of the additional mass by a shaker and accelerometer on the natural frequency is investigated using the Rayleigh-Ritz method. A cutting test is conducted to verify whether appropriate cutting conditions can be set from the measurement results of the on-machine dynamic stiffness measurement. The natural frequency difference between the on-machine measurement method and impact test was caused by the additional mass by the contact of the piezoelectric actuator. When the additional mass is 5% or less of the workpiece equivalent mass, the difference in natural frequency was as small as 10 Hz or less. In the cutting test using the workpiece with the sufficiently large mass, the compliance between the on-machine measurement method and impact test was comparable. When the forced vibration was small, the stability limit was correctly estimated from the on-machine dynamic stiffness measurement.

Keywords: Measuring instrument; stiffness; workpiece; piezoelectric; evaluation;

1. Introduction

Machining of thin-walled workpieces is a popular issue due to its static and dynamic displacements during machining. It is required to set machining conditions sufficiently considering the vibration characteristics of the workpiece. Thus, the dynamic stiffness measurement of workpiece is important in machining process for improving machining efficiency and accuracy [1, 2].

The dynamic stiffness measurement requires an excitation device with a force sensor and an instrument to measure the response of the workpiece. In general, an accelerometer is used for the response sensor. For example, in the modal analysis of a fan case of an aircraft engine, the measurement is performed using an impact hammer and accelerometers [3]. However, it takes a lot of time and manpower for the sensor set up because of a large number of accelerometer attachment points. Moreover, the reliability of the measurement result depends on the operator's skill. Since the vibration characteristics of the workpiece changes with the change in the workpiece thickness during machining process from rough machining to finishing, it is necessary to repeat the same test. An automatic on-machine system to measure the dynamic stiffness of the workpiece enables a reliable and frequent measurement.

Therefore, we proposed Displacement Sensorless Piezo Excitation method for stiffness measurement (DSPE method)

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[4]. DSPE method is a dynamic stiffness measurement method which does not require a response sensor on the workpiece. The displacement of the workpiece is estimated using an equivalent mechanical model of the piezoelectric element. A prototype of the on-machine measurement device has been developed. The obtained dynamic compliance was comparable to that with the Piezo Excitation method (hereinafter referred to as PE method) using the same prototype, which is a conventional dynamic stiffness measurement method [5, 6]. However, the compliance measured with impact test and that with PE method differed in natural frequency of the workpiece.

In this paper, the developed on-machine measurement device is evaluated. The cause of the difference in natural frequency between PE method and impact test is investigated. The measurement system model is constructed using the Rayleigh-Ritz method. The influence of the additional mass by a shaker and accelerometer on the natural frequency is investigated. In addition, a cutting test is conducted to verify whether appropriate cutting conditions can be set from the measurement results of the on-machine dynamic stiffness measurement.

2. Dynamic stiffness measuring device and measuring method

Figure 1 shows the experimental setup of PE method. An excitation device is attached to the spindle using the tool holder. The shaker is composed of a cylindrical jig, a piezoelectric actuator, and a force sensor.

In the measurement, a free end of a plate workpiece is excited with the shaker. The excitation force is measured with a force sensor (Kistler) attached to the shaker. The acceleration of the shaker and workpiece is measured with attached accelerometers (PCB Piezotronics). From the measured excitation force and the acceleration of the workpiece, the accelerance is obtained by H_1 estimation. The measured accelerance is integrated to obtain the compliance. The accelerometer attached to the shaker is used to confirm that the vibration of the shaker is sufficiently smaller than that of the workpiece. A metal shield type piezoelectric actuator (TOKIN) is used for the actuator.

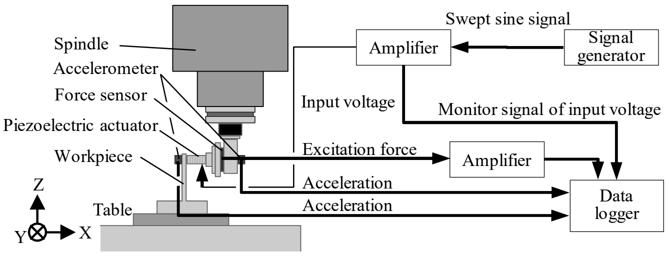

Fig.1. Experimental setup of PE method

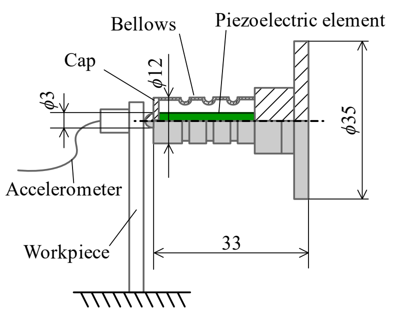

Fig.2. Schematic view of piezoelectric actuator

3. Comparison between model by energy method and experiment

3.1. Decrease of natural frequency by additional mass

In our previous research [4], the dynamic stiffness measurement of cantilevered plates was conducted. The natural frequency of the primary bending mode measured with PE method was lower than that measured with the impact test. In this section, the cause of this difference is investigated.

In PE method, the excitation is performed while the tip of the actuator contacts with the workpiece. In such case, the mass of the shaker can affect the measurement result [4]. The natural frequency of the workpiece can be decreased by the additional mass on the free end. When the additional mass is constant, the decrease rate of the natural frequency should vary depending on the equivalent mass of the workpiece.

Therefore, the natural frequencies are calculated by the Rayleigh-Ritz method for three types of workpieces with different masses. The excitation experiment is conducted to compare the experimental result and analytical calculation. It verifies that the mass of the actuator tip acts as an additional mass and causes the difference between the natural frequency of PE method and impact test.

3.2. Vibration analysis of cantilever beam by Rayleigh-Ritz method

Figure 2 shows a schematic view of the measurement in PE method. A cantilever plate is excited in this experiment. The plate is approximated to a cantilever beam for simplicity because this paper focuses on the first order bending vibration. The piezoelectric actuator has a multilayer piezoelectric element covered by the metal case. The metal case consists of a stainless steel cap and bellows. The mass of an accelerometer, the piezoelectric element, the cap, and the bellows acts as an additional mass on the cantilever in the model of PE method. In the model of the impact test, only the mass of the accelerometer is taken into account.

The mass of the cap, bellows and piezoelectric element are 2.97 g, 0.99 g and 4.80 g, respectively. The size of the piezoelectric element is 5 mm × 5 mm × 20 mm, and the maximum displacement is about 17 μm. The sizes of three workpieces A to C excluding the fixed part are as follows; A: 100 mm × 100 mm × 5 mm, B: 70 mm × 100 mm × 5 mm, C: 65 mm × 50 mm × 5 mm. The workpiece material is carbon steel S50C. The accelerometer position and the excitation position are the center of the plate and 6 mm apart from the free end. The mass of the accelerometer is 7.74 g.

When the additional mass m_{add} is attached to the workpiece and the distance between the mass and workpiece free end is x_0, the maximum kinetic energy T is expressed by the following equation;

$$T = \frac{1}{2}\omega^2 \int_0^l \rho A\, y^2 dx + \frac{1}{2} m_{add}\omega^2 y(x_0)^2 + \frac{1}{2}\omega^2 \int_0^{l_p} \rho_p\, A_p\, u^2 dx \tag{1}$$

where E is the elastic modulus of the workpiece, I is the second moment of inertia, l is the length of the beam, ρ is the density, A is the cross sectional area, l_p the length of the piezoelectric element, ρ_p is the density of the piezoelectric element, A_p the cross sectional area of the piezoelectric element, u is the displacement of the tip. The displacement of the tip equals to the displacement of the workpiece in the result of a preliminary experiment. Thus, the deflection of the cantilever $y = u$.

Also, the maximum strain energy V is expressed by the following equation;

$$V = \frac{1}{2}\int_0^l EI\left(\frac{d^2y}{dx^2}\right)^2 dx \qquad (2)$$

When the concentrated load P acts on the free end, the static deflection y is calculated as follows;

$$y = \frac{Pl^3}{3EI}\left\{1 - \frac{3}{2}\left(\frac{x}{l}\right) + \frac{1}{2}\left(\frac{x}{l}\right)^3\right\} \qquad (3)$$

Eq.(3) is substituted into Eq.(1) and (2). The natural frequency ω is obtained when $T = V$.

3.3. Experimental method and results

In the excitation experiment, a sinusoidal voltage was input to the piezoelectric actuator. The preload of 100 N was given to the workpiece and the piezoelectric actuator using the feed of the machine tool. The frequency of the input voltage was swept from 1 Hz to 1200 Hz for 30 s. The sampling frequency was 15 kHz. The impact test was carried out using an impact hammer (PCB Piezotronics) and the accelerometer.

Table 1 shows the natural frequencies obtained from the analytical calculation and excitation experiments for each workpiece. Figure 3 shows the relationship between the natural frequency difference between the impact test and PE method and the equivalent mass of the workpiece. In both of experimental and analytical results, a larger frequency difference between the impact test and PE method is observed for a smaller equivalent mass of the workpiece. This result indicates that the tip of the actuator acts as an additional mass.

The additional mass in the analytical calculation is 4.90 g in PE method. When the additional mass is 5% or less of the equivalent mass, the difference in natural frequency is as small as 10 Hz or less. Therefore, in this paper, the equivalent mass is sufficiently large and the influence of additional mass is small in case of Workpiece A. Figure 4 shows the measured compliance of Workpiece A. It shows that similar results are obtained by PE method and impact test. Furthermore, the compliance measured with DSPE method corresponds well with that of the PE method. Therefore, DSPE method is comparable to the conventional shaker test and impact test.

Table 1. Natural frequency of workpieces

	Workpiece	A	B	C
Measured frequency (Hz)	Impact test	284	820	889
	PE method	286	790	835
Calculated frequency (Hz)	Impact test	283	814	960
	PE method	279	791	911

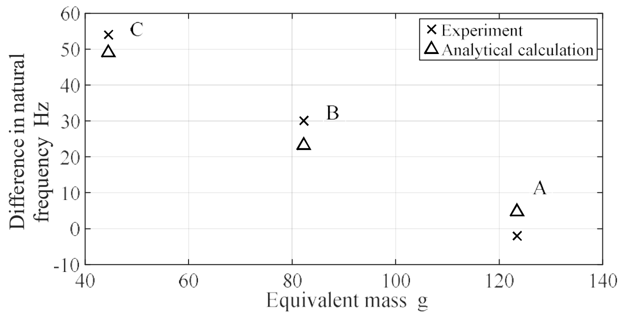

Fig.3. Relationship between equivalent mass and natural frequency difference between PE method and impact test

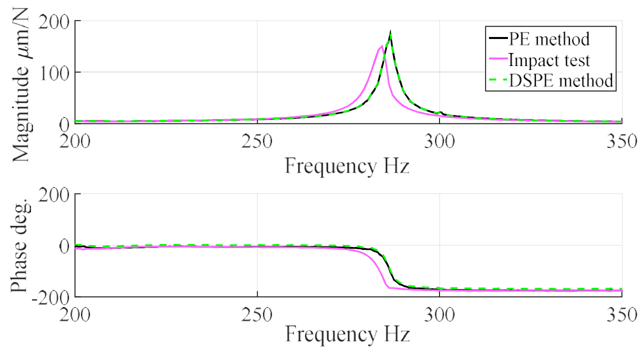

Fig.4. Compliance of workpiece for 200-350 Hz.

4. Cutting test

The validity of the on-machine dynamic stiffness measurement in the decision of cutting conditions is verified. From the dynamic stiffness measured with PE method and impact test, a stability limit is calculated by the method proposed by Y. Altintas, et al [8]. In addition, a cutting test was carried out using Workpiece A to obtain the stability limit and compared with the calculation result.

Figure 5 shows the setup of the cutting test. A side cutting using an end mill was carried out. The axial depth of cut was continuously varied from 5 mm to 0 mm. Table 2 shows cutting conditions. The spindle rotation speed was set under each condition in the range of 1000 - 3200 min^{-1}.

The vibration during cutting was measured with a 1-axis accelerometer to determine the stability limit. The workpiece displacement was obtained by integration. The amplitude spectrum of the workpiece displacement is obtained every 0.1 s during cutting. When the amplitude of the components at frequencies other than the tooth passing frequency and its harmonics exceeds a threshold, the axial depth of cut at the moment is determined to the stability limit. The threshold was set to 5 μm in this experiment.

Figure 6 shows the experimental and analytical stability limit diagrams. Based on the results of preliminary experiments, the cutting force coefficients K_t was set as 2588 MPa and the component of force ratio K_r was set as 0.48. The qualitative trends of the stability analysis result and the cutting experiment result are similar, but the critical depth of cut has a difference. In this study, the stability analysis focused only on the bending mode of the workpiece. An analysis considering the influence of the torsional mode is required. Additionally, since the cutting test was conducted with the low stiffness workpiece and the small radial immersion of cut, the cutting process was intermittent. The difference of the stability limit can be caused by the change of the tool-workpiece dynamic characteristic depending on the tool contact condition.

Table 2. Experimental condition of cutting test

Tool	Type	Square endmill
	Material	Carbide
	Number of tooth	4
	Diameter mm	14
Milling direction		Down cut
Coolant		Dry
Radial depth of cut mm		0.1
Axial depth of cut mm		0-5
Feed rate mm/min^{-1}		400-1300
Spindle Speed min^{-1}		1000-3400
Feed per tooth mm/tooth		0.1

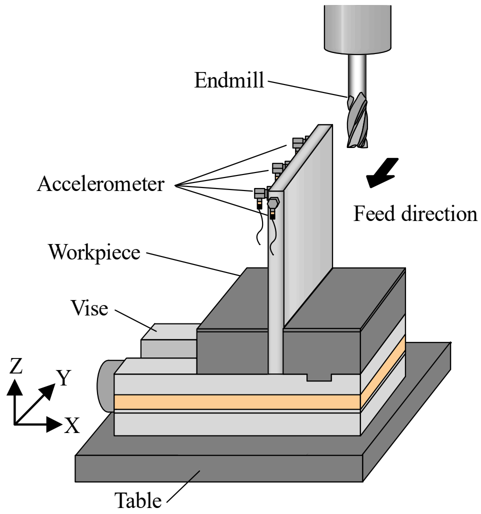

Fig.5. Experimental setup for cutting test

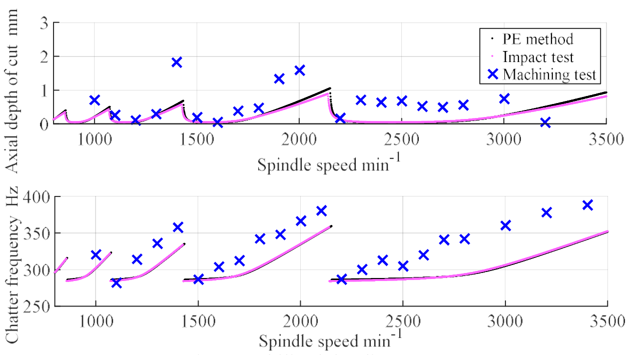

Fig.6. Stability lobe diagram

5. Conclusion

The developed on-machine measurement device of the workpiece dynamic stiffness is evaluated. The cause of the difference in the natural frequency between PE method and impact test is investigated. The validity of the on-machine dynamic stiffness measurement in the decision of cutting conditions is verified. From this study, the following conclusions are obtained.

(1) The natural frequency of the system was calculated using Rayleigh-Ritz method. The comparison between the calculation and experiment showed that the natural frequency difference between PE method and impact test is caused by the additional mass by the contact of the piezoelectric actuator. When the additional mass is 5% or less of the workpiece equivalent mass, the difference in natural frequency is as small as 10 Hz or less.

(2) In the cutting test using the workpiece with the sufficiently large mass, the compliance between PE method and impact test is comparable. The stability limit was estimated from the on-machine dynamic stiffness measurement.

Acknowledgements

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